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High Energy Flywheel Containment Evaluation

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Introduction

A flywheel testing facility is being constructed at the NASA Glenn Research Center. This facility is to be used for life cycle testing of various flywheel rotors. The lifecycle testing consists of spinning a rotor from a low RPM (~ 20,000) to a high RPM (~60,000) and then back to the low RPM. This spin cycle will model that which the rotor will see during use. To simulate the lifetime of the rotor, the spin cycle will be performed tens of thousands of times. A typical life cycle spin test is expected to last 6 months. During this time the rotor will be spun through a cycle every 5 minutes. The test will run continuously for the 6 month period barring a flywheel failure. Since it is not reasonable to have the surrounding area evacuated of personnel for the duration of the testing, the flywheel facility has to be designed to withstand a flywheel rotor failure and insure that there is no danger to any personnel in the adjacent buildings or surrounding areas.

In order to determine if the facility can safely contain a flywheel rotor failure an analysis of the facility in conjunction with possible flywheel failure modes was performed. This analysis is intended as a worst case evaluation of the burst liner and vacuum tank's ability to contain a failure. The test chamber consists of a cylindrical stainless steel vacuum tank, two outer steel containment rings and a stainless steel burst liner. The stainless steel used is annealed 302, which has an ultimate strength of 620 MPa (90,000 psi). A diagram of the vacuum tank configuration is shown in figure 1. The vacuum tank and air turbine will be located below ground in a pit. The tank is secured in the pit with 0.3 m (12 in) of cement along the base and the remaining portion of the tank is surrounded by gravel up to the access ports. A 590 kg (1300 lb) bulkhead is placed on top of the pit during operation and the complete facility is housed within a concrete structure which has 7.5 cm (3 in) thick walls. A cutaway of the facility is shown in figure 2.

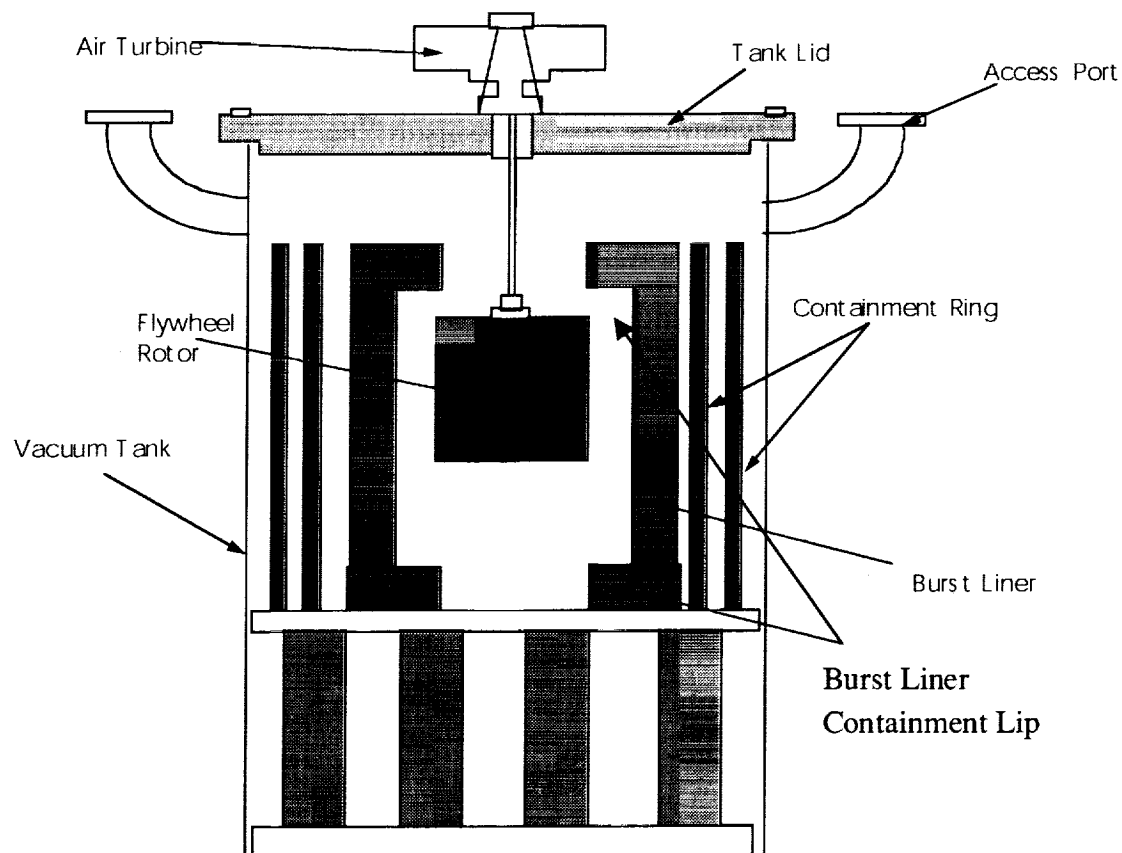


Figure 1 Vacuum Tank Layout

The specifications of the flywheel rotor used in this analysis are as follows. The rotor was constructed of a wound composite outer surface with a titanium metal hub. The outer composite ring was 2.8 cm (1.1") thick. The rotor had an outer diameter of 26.7 cm (10.5") and a length of 33 cm (13"). The density (ρ_r) used for the composite outer ring was 1608 kg/m³ (0.058 lb/in³).

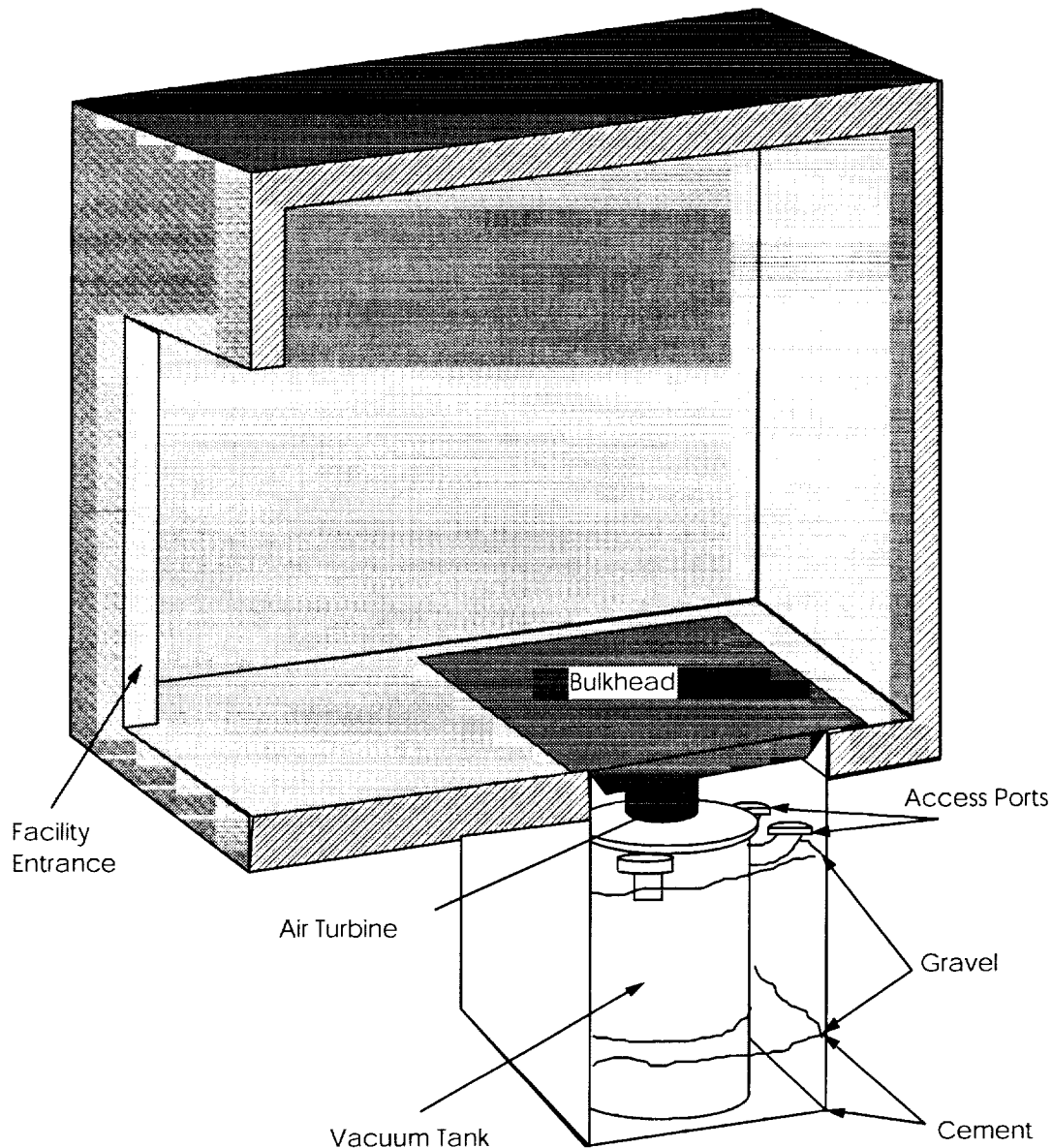


Figure 2 Flywheel Test Facility Layout

The analysis performed was a first order calculation to determine if the burst liner and vacuum tank could withstand a failure of the flywheel. The first order calculations were deemed sufficient since the impact assumptions represented a worst-case situation for each of the failure mechanisms examined. The rationale is that if the facility can withstand the worst-case failure, for each of the failure mechanisms examined, then any other type of failure should be contained. Three failure modes for the flywheel were examined. The first was a tri-burst in which the outer ring breaks into three segments and each segment impacts the outer wall of the burst liner. The second was the fragmentation of the outer ring of the flywheel producing a uniform pressure on the inner surface of the burst liner. A third mechanism examined was a tri burst failure resulting in the outer ring deflecting upward and impacting the containment lip on the burst liner. A variation on

this third mechanism was the deflection and subsequent fragmentation of the rotor outer disc. For this situation it was assumed that the rotor particles were uniformly distributed and impacted both the burst liner lip and vacuum tank lid

Analysis

Tri-burst Impact with Burst Liner Wall

The tri-burst impact assumes that the rotor breaks into three segments and that each segment impacts the burst liner intact. All of the energy in the segment is converted to the force of impact. No energy dissipation is assumed from material deformation or heating. The area of each impact is equal to the surface area of 1/3 of the rotor. Since each of the impact events is isolated the maximum stresses will occur within the impact area and will be the same for each. The area of impact (A_i) is given by:

$$A_i = 2\pi r_i h / 3 = 2\pi * 0.1335 * 0.33 / 3 = 0.092 \text{ m}^2 (142.9 \text{ in}^2) \quad [1]$$

Where r_i is the rotor radius and h is the rotor height. The pressure exerted on this surface is due to the force of impact (F). This force is calculated as follows;

$$F = M_{ir} V_i / t_i = 20.82\text{E}6 \text{ N} (4.68\text{E}6 \text{ lb}) \quad [2]$$

Where M_{ir} is the mass of the rotor segment, V_i is the translated linear velocity of the rotor segment and t_i is the impact duration. The mass of the rotor segment is given by the volume of the segment and the material density (ρ_r).

$$M_{ir} = \pi (r_o^2 - r_i^2) h \rho_r / 3 = 1.96 \text{ kg} (4.32 \text{ lb}) \quad [3]$$

The impact duration is assumed to be 75 μsec . This assumption was based on information obtained from reference 1. The translated linear velocity is determined by the following equations.

$$V_i = \sqrt{(E_r / M_{ir})} = 795 \text{ m/s} (2609 \text{ ft/s}) \quad [4]$$

Where the rotational energy in the flywheel outer ring (E_r) is given by the following with an assumed RPM of 60,000;

$$E_r = I \omega^2 / 2 = 1.86\text{E}6 \text{ J} \quad [5]$$

$$I = \pi h \rho_r ((r_o)^4 - (r_i)^4) / 2 = 0.094 \text{ m}^4 (321.82 \text{ in}^4) \quad [6]$$

$$\omega = 2\pi \text{ RPM} / 60 = 6283.2 \text{ radians/second} \quad [7]$$

From the force given in equation 2 and the area in equation 1 the pressure exerted on the burst liner wall can be calculated.

$$P = F / A_i = 2.26\text{E}8 \text{ Pa} (32,752 \text{ psi}) \quad [8]$$

Utilizing this pressure the maximum stress in the burst liner wall can be calculated. Because the pressure is acting on only a segment of the burst liner, the stress was calculated by assuming that the stress distribution was similar to that of a rectangular surface supported along its edges (this rectangular surface is the same as the height, h, and thickness, t, of the rotor by 1/3 its circumference, c). Based on this assumption the maximum stress is given by the following equation².

$$\sigma_{\max} = h^2 (c/3)^2 P / (2 (h^2 + (c/3)^2) t^2) = 4.68\text{E}8 \text{ Pa (67,832 psi)} \quad [9]$$

This maximum stress is less than the ultimate strength of the burst liner and therefore should not cause it to fail under the assumed loading.

Outer Ring Disintegration and Impact with Burst Liner Wall

The outer ring disintegration assumes that the outer composite ring of the flywheel breaks apart and strikes the containment wall. For this type of failure it is assumed that the outer ring material is evenly distributed and extends out horizontally from the flywheel. There is no vertical distribution of the flywheel material. This band of material strikes the burst liner at the same instant producing a uniform pressure band along the liner's inner wall. The width of this band is equal to the height of the rotor. In this analysis it was assumed that no energy was dissipated due to heating or particle deformation. The area of the impact is given by the following equation where Ri is the radius of the burst liner's inner wall.

$$A_i = h \, 2\pi \, R_i = .546 \text{ m}^2 \text{ (847 in}^2\text{)} \quad [10]$$

The force of the impact is calculated in the same manner as that shown in the tri-burst analysis given above. However since the force of impact used in the tri-burst was for one third of the rotor outer ring this force has to be multiplied by three.

$$F = 6.25\text{E}7 \text{ N (14.05E6 lb)} \quad [11]$$

The pressure (P) on the burst liner wall is therefore this force times the area of impact. This is represented by equation 8.

$$P = 1.14\text{E}8 \text{ Pa (16,574 psi)} \quad [12]$$

From this pressure the maximum stress within the burst liner wall can be calculated. This stress for a thick walled cylinder is given by the following relation². Where Ro is the outer radius of the burst liner and Po is the external pressure. Since the burst liner is within a vacuum tank the external pressure can be neglected (Po = 0).

$$\sigma_{\max} = P ((R_i^2 + R_o^2) - 2 P_o R_o^2) / (R_o^2 - R_i^2) = 3.54\text{E}8 \text{ Pa (51,346 psi)} \quad [13]$$

This maximum stress is less than the ultimate strength of the burst liner and therefore should not cause it to fail under the assumed loading.

Containment Lip Impact

The impact of the flywheel outer ring onto the containment lip of the burst liner was analyzed. This analysis was done for two situations. The first assumed that the outer ring of the flywheel fragmented, impacted the burst liner wall and was then deflected vertically (equal amounts upwards and downwards) along the wall and impacted the upper and lower containment lips uniformly. The second assumed the flywheel fragmented, impacted the burst liner wall, was deflected vertically (equal amounts upwards and downwards) and the fragments were evenly distributed over the end surfaces of the burst liner. In this second case some of the flywheel particles escape the burst liner and impact the vacuum tank lid. For both cases it was assumed that no energy was lost during the impact with the burst liner wall, only the direction of the particles were changed to vertical. It was also assumed that one half of the particles were deflected upward and the other half downward. The determination of the maximum shear stress within the lip is calculated by determining the force acting on the lip. Based on the assumptions made this force will be uniform along the lip surface. Since the lip is basically a cantilevered surface, the maximum stress will occur at the base of the lip nearest to the burst liner wall.

For the first case the pressure exerted on the lip is the force, given in equation 11, divided by the area of the lip. This area is given by the following relation, where R_{il} is the inner radius of the lip, 0.187m (7.375 in) and R_{ol} is the outer radius of the lip, 0.264 m (10.375 in).

$$A_i = \pi (R_{ol}^2 - R_{il}^2) = 0.108 \text{ m}^2 (167 \text{ in}^2) \quad [14]$$

$$P = F / A_i = 2.89\text{E}8 \text{ Pa} (41,979 \text{ psi}) \quad [15]$$

Where F is the force of impact of one half of the rotor outer ring mass. This is equal to one half of the force given in equation 11, $5.7\text{E}7 \text{ Pa}$ ($7.025\text{E}6 \text{ lb}$). The maximum stress will be due to the bending moment caused by this pressure on the lip. The lip of the burst liner can be thought of as a circular plate with a hole in the center. This plate is rigidly fixed along its edge to the burst liner walls. Using this analogy an equation for the maximum stress in the burst liner lip can be obtained from reference 3. This equation is given as follows, where the coefficient k is determined empirically based on the ratio of the plate radius and hole radius. For this case k is approximately 0.259. The thickness, t , of the burst liner lip is 12.7 cm (5 in).

$$\sigma_{\max} = k P R_{ol}^2 / t^2 = 3.23\text{E}8 \text{ Pa} (46,831 \text{ psi}) \quad [16]$$

This value for the maximum stress in the burst liner lip is less than the ultimate strength of the burst liner material. Therefore the lip should not fail under the flywheel failure mode described above.

The second vertical fragmentation case assumes that the evenly distributed particles impact both the burst liner lip and the lid of the vacuum chamber. The stress on the lip will be less than that of the previous variation, where all the particles impacted the lip. Therefore the integrity of the burst liner lip does not have to be reevaluated for this case. The particles, which escape the burst liner, will impact the lid with a force proportional to the exit area of the burst liner. This area ratio is represented by the square of the ratio of the inner and outer radius of the burst liner lip.

$$R_{il}^2 / R_{ol}^2 = 0.505 \quad [17]$$

This indicates that approximately 50% of the particles pass through the opening in the burst liner and impact the lid. The force of this impact is given by the force generated from one half the rotor mass, $3.12E7$ N ($7.02E6$ lb), multiplied by the fraction given above.

$$F = 1.58E7 \text{ N (3.55E6 lb)} \quad [18]$$

The lid is a stainless steel plate with a port in the center for the turbine shaft to pass through. It is secured to the vacuum tank with 24 evenly spaced bolts, 1.9 cm (.75 in) in diameter. First it must be determined if the bolts can withstand the impact force given by equation 18. It is assumed that if the bolts fail they will fail in tension along the shaft of the bolt above the threads. Using this assumption the force required for a bolt to fail is based on its cross sectional area and the ultimate strength of the material. For the stainless steel bolts being used this ultimate strength, σ_{ult} , is given as $6.21E8$ Pa (90,000 psi). The bolt cross sectional area, A_c , is given by the following equation where R_b is the bolt radius.

$$A_c = \pi R_b^2 = 2.85 \text{ cm}^2 (0.442 \text{ in}^2) \quad [19]$$

The force required to break all of the bolts is given by the following equation.

$$F = 24 \sigma_{ult} A_c = 4.24E6 \text{ N (9.5E5 lb)} \quad [20]$$

Since the force given in equation 20 is less than that given in equation 18, the bolt strength is not sufficient to hold the lid onto the tank for this type of worst-case impact. Based on this result, lid locks were constructed to secure the tank lid to the tank. Diagrams of these lid locks are given in figures 3 and 4. Four lid locks were constructed. These locks will be positioned symmetrically around the tank. The locks are channel shaped pieces of metal, which would slide over the tank lid and under a lip on the tank.

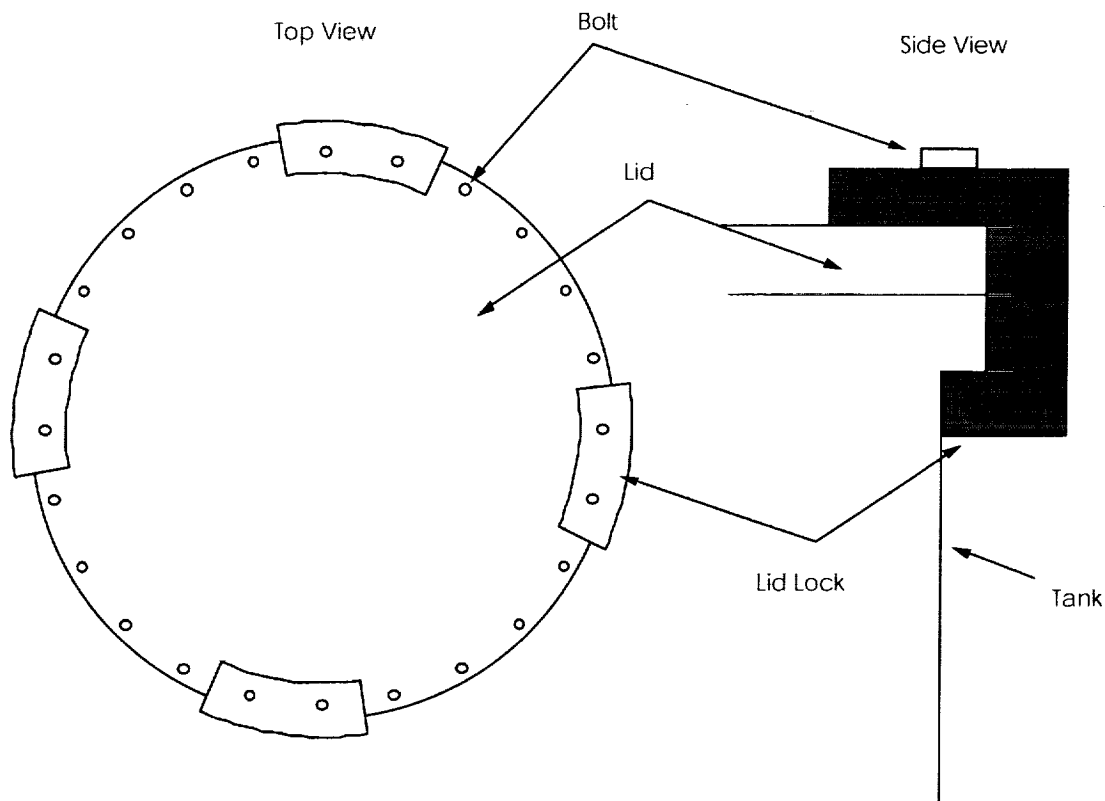


Figure 3 Lid Lock Positioning

The force required to break the lid locks is their cross sectional area, 110.5 cm^2 (17.13 in^2), multiplied by the ultimate strength of the material, which is $8.27\text{E}8 \text{ Pa}$ ($120,000 \text{ psi}$). This force is given by the following equation.

$$F = 4 (0.011) (8.27\text{E}8) = 3.655\text{E}7 \text{ N} (8.22\text{E}6 \text{ lb}) \quad [21]$$

This force, given above, is much larger than the force generated by the impact given in equation 18 and therefore indicates that if the lid locks are used the lid will remain on the tank in the event of the assumed type of impact.

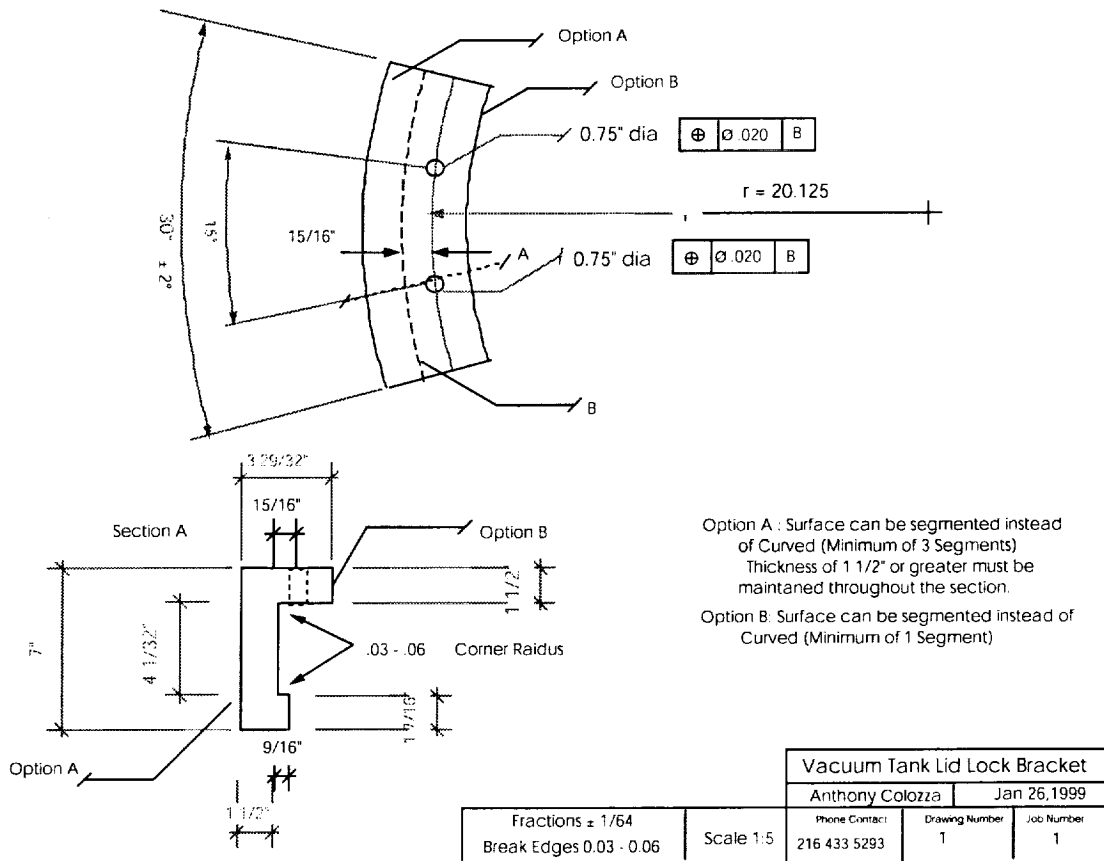


Figure 4 Lid Lock Diagram

Even though the lid attachment to the vacuum tank can withstand the estimated forces, the lid itself must be capable of withstanding these forces. In order to assess this the maximum stress in the lid must be calculated. The resulting pressure on the lid due to the force of impact is given by the area of impact divided by the force of impact. The area of impact is based on the inner diameter of the burst liner lip. This area and subsequent pressure are given by the following equations.

$$A_i = \pi R_{il}^2 = 0.11 \text{ m}^2 (170.9 \text{ in}^2) \quad [22]$$

$$P = F / A_i = 1.43\text{E}8 \text{ Pa} (20,767 \text{ psi}) \quad [23]$$

Utilizing this pressure the maximum stress in the lid can be calculated³. This stress assumes that the lid is fixed along its edge. The lid thickness, t , is 7.6 cm (3 in).

$$\sigma_{\max} = 0.75 P (R_{il} / t)^2 = 6.49\text{E}8 \text{ Pa} (94,126 \text{ psi}) \quad [24]$$

This above relation is based partly on empirical data and assumes an “average – thickness” plate in which the flexure stress predominates. This equation does not account for stress redistribution due to local yielding of the lid. In practice this type of local yielding will reduce the maximum stress seen on the plate. Therefore the result given above is conservative. The maximum stress given in equation 24 is greater than the yield strength of the lid material. This suggests that the lid will fail under the burst conditions specified above. However, since the case being analyzed is a worst-case type of failure which does not take into account energy absorbing aspects of the impact such as particle deformation, heating or energy lost through deflection. The inclusion of these items, through a more detailed analysis, should reduce the force of impact of the flywheel particles on the lid. Considering these factors along with the conservative nature of the equation used for the maximum stress calculation indicates that there is considerable margin built into the analysis. Since the maximum stress is within 4.5% of the ultimate strength of the material it is believed that this calculated failure of the lid would not occur during an actual failure of the rotor. In order to better resolve this issue a more detailed analysis would need to be done taking into account the impact factors listed above.

The tank lid has a 7.67 cm (3.02 in) diameter port at its center. This port is for the mounting of the air turbine. The air turbine is mounted with four 0.95 cm (3/8 in) diameter bolts. The force on the turbine during the failure described above is based on the impact of the flywheel particles that pass through the turbine mounting port. The pressure through this port is the same as that given in equation 23. Based on this pressure the force on each of the four bolts is calculated below, where A_{tp} is the area of the turbine port, 46.21 cm^2 (7.16 in^2).

$$F = P A_{tp} / 4 = 165,416 \text{ N (37,189 lb)} \quad [25]$$

The maximum force the bolts can withstand is given by the ultimate strength (σ_{ult}) of the bolt material and the bolts cross sectional area (A_b). For stainless steel bolts the ultimate strength is $6.21\text{E}8 \text{ Pa}$ (90,000 psi) and the cross sectional area for each bolt is 0.713 cm^2 (0.11 in^2).

$$F = \sigma_{ult} A_b = 44,218 \text{ N (9,940 lb)} \quad [26]$$

Based on this the turbine bolts will not be able to withstand the force given in equation 28. Therefore a separate plate will need to be mounted on the inside of the vacuum tank lid in order to reduce the turbine port opening. A 2.54 cm (1 in) thick 304 stainless steel plate with a 2.54 cm (1 in) hole in the center was chosen. The hole size is sufficient to allow the turbine shaft to pass through and into the vacuum tank. The stress on this plate is given by the following equation³ where the pressure, P , on the plate is given by equation 23, the radius of the plate (R_{po}) is equal to the port opening 7.67 cm (3.02 in) and t is the plate thickness.

$$\sigma_{max} = 0.654 P (R_{po} / t)^2 = 2.14\text{E}8 \text{ Pa (30,967 psi)} \quad [27]$$

This plate should not fail since the maximum stress is below the plate material's ultimate strength.

With this plate in place the force impacting the turbine can be calculated as follows, where the area of the port opening, A_{po} , is 5.067 cm (0.785 in).

$$F = P A_{po} = 72,548 \text{ N (16,310 lb)} \quad [28]$$

This results in a force per turbine bolt of 18,137 N (4,077 lb) which is well below the ultimate strength for these bolts. Therefore the turbine will stay attached to the vacuum tank lid during the type of flywheel failure described above.

Titanium Nitrogen Interaction

Titanium and nitrogen can react and burn under specific conditions. This reaction potential may be a concern in the spin pit facility since a number of the flywheel rotors have titanium components and nitrogen is used in the operation of the facility. Nitrogen is presently used as the backfill gas during shutdown and emergency operations and it may potentially be used as a mechanism to heat the rotor during testing.

Titanium is a flammable metal of the alkali group. It burns in oxygen at 610° C (1130°F) to form titanium dioxide and it burns in nitrogen at 800° C (1472°F) to form titanium nitride (TiN). It can burn in air and is the only element that can burn in nitrogen. Once ignition occurs titanium will burn with great intensity. While burning the material is water reactive and an explosion will result if water is added to the flame. A potential source for a rotor fire would be the inadvertent backfill or venting of the chamber during rotor operation. The friction due to the presence of the nitrogen gas or air would cause the rotor to heat up potentially to the point of ignition.

Combustible dusts are a finely divided (particulate) form of a solid material that will burn when mixed, in the correct proportion, with a reactive gas. Because of titanium's ability to react with oxygen and nitrogen there is the potential of a dust explosion under specific conditions. This can occur if, by some mechanism, titanium dust is generated by the rotor during failure. If the tank, in the presence of this dust, is then backfilled with nitrogen an explosion may result. This explosion is actually a fast moving fire that propagates from particle to particle in a fraction of a second. This rapid release in energy produces a tremendous increase in pressure and heat. If the dust particles are small and distributed throughout the chamber volume, the explosion can be triggered by a source as small as a static electric spark.

Titanium is classified as being a severe dust explosion hazard. Under dust explosion conditions it has one of the highest rates of pressure rise (greater than 68.95E6 Pa (10,000 psi) per second) of any explosive dust material and produces a maximum explosive pressure of 551.6E3 Pa (80 psig). The minimum concentration necessary for a dust explosion is 0.045 kg/m³ (0.045 oz/ft³).

Conclusion

Based on this worst-case analysis the burst liner and vacuum tank should be capable of withstanding any of the types of flywheel failures examined. The vacuum tank lid has the greatest potential for failure of all the components which were examined. However, as discussed previously the nature of this basic analysis lends itself to being highly conservative. Because of this it is felt that the vacuum tank lid will be capable of containing a flywheel failure without rupturing. In order to address this issue fully a more detailed analysis would need to be performed which takes into account the various energy absorbing mechanism, such as particle heating and deformation, which are ignored in this basic analysis.

Since titanium does react with nitrogen the following two points must be considered when evaluating the safety of the spin pit facility. Under failure conditions the temperature of the titanium part of any rotor must not be capable of exceeding 800° C. The possibility of generating titanium dust during a failure of the rotor must be eliminated or kept below the critical 0.045 kg/m^3 (0.045 oz/ft^3) level.

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2. Hicks Tyler G. (editor), "Standard Handbook of Engineering Calculations", McGraw Hill Book Company, New York, 1972.
3. Baumeister, Theodore (editor), "Mechanical Engineers Handbook", McGraw- Hill Book Company, New York, 1958.

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